Optimal Terminal Box Control for Single Duct Air-Handling Units

YoungHum Cho, Jiri Vondal, Gang Wang and Mingsheng Liu
University of Nebraska – Lincoln
yhcho@mail.unomaha.edu

Abstract: Terminal boxes maintain room temperature by modulating supply air temperature and airflow in building HVAC systems. Terminal boxes with conventional control sequences often supply inadequate airflow to a conditioned space, resulting in occupant discomfort, or provide excessive airflow that wastes significant reheat energy. In this study, an optimal terminal box airflow control sequence was developed to improve indoor ventilation and reduce energy consumption. The developed control sequence was applied in an office building air conditioning system. Improvements in indoor thermal comfort and energy reduction were verified through measurement. The results show that the optimal control sequence can stably maintain thermal environment, satisfy comfort standards and reduce energy consumption compared to the conventional control sequence.

1. INTRODUCTION

Terminal boxes control space conditions in variable air volume (VAV) air-handling unit (AHU) systems. Terminal boxes either modulate airflow with a control damper or adjust discharge air temperature with a reheat coil. Terminal boxes with conventional control sequences may cause occupant discomfort if airflow is too low or waste excessive reheat energy if airflow is too high. The terminal box minimum airflow set point is related to the terminal box airflow. Terminal boxes will have significant simultaneous heating and cooling and AHUs will consume more fan power if the minimum airflow is higher than required. On the other hand, conditioned space will have indoor air quality (IAQ) problems with less air circulation if the minimum airflow is less than required. Low minimum airflow is a primary cause of IAQ complaints. Therefore, minimum airflow optimization should be studied.

Taylor et al. (2004) discussed the minimum airflow set point of terminal boxes. The actual controllable minimum airflow set point is usually different than the box manufacturer's recommended minimum airflow set point for each box size and for each standard control option. An improved new terminal VAV box zone control was suggested by Hartman et al. (2003). The terminal box damper is typically bounded by presetting minimum and maximum airflow set points and the minimum airflow set point is usually based on outdoor air ventilation requirements. Building HVAC equipment routinely fails to satisfy performance expectations due, in part, to control problems related to incorrect sequencing logic, such as minimum and maximum airflow set points (Schein et al. 2005).

Indoor thermal comfort is related to the vertical temperature difference (ASHRAE 1992 and Olesen 2002). The vertical distribution is kept lower than the value, below 3ºC (5.4 ºF) between head and ankles (1.1m and 0.1 m above the floor). CO_2 concentration is a key parameter to evaluate the indoor air quality. In ASHRAE standard 62, the indoor CO_2 level can be accurately maintained at approximately 700 ppm above the outdoor CO_2 level.

The objective of this study is to optimize the minimum airflow set point to improve thermal environment and save energy consumption. This paper presents problems with conventional terminal box controls, develops a procedure to determine optimal minimum airflow set point to solve these problems, and applies the optimal minimum airflow to an actual AHU system. The indoor thermal comfort and the terminal box energy consumption of conventional and improved control sequences were compared using measured data.

2. TERMINAL BOX CONTROL

Generally, a terminal box without reheat coil is used in interior zones. On a call for cooling, airflow will increase as the damper opens towards the maximum airflow set point to satisfy the room air temperature set point. On a call for less cooling, airflow will decrease as the damper closes towards the minimum airflow set point to satisfy the room air temperature set point. On the other hand, the terminal box in exterior zones has a reheat coil. The control sequence of the exterior zone box adds a reheat coil valve control sequence(?) based on that of the interior zone terminal box. With a further call for heat, airflow drops to the minimum airflow set point and the reheat coil valve is modulated to maintain the room air temperature heating set point. Figure 1 presents a schematic of a VAV terminal box with a reheat coil. The single-duct VAV terminal box with a reheat coil consists of a controller, an actuator, a damper, a flow sensor and a reheat coil valve. Figure 2 shows its control sequence.

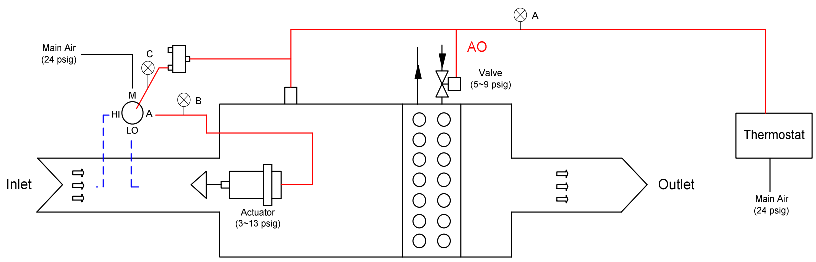


Fig. 1: Schematic of a single-duct VAV terminal ox with a reheat coil

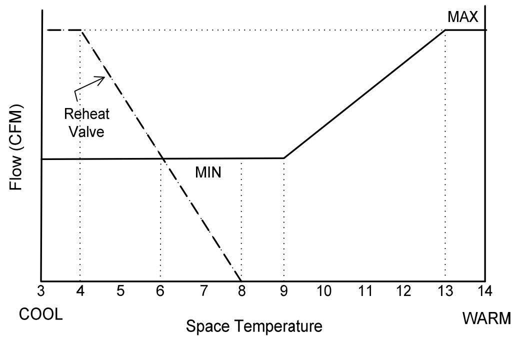


Fig.2: Control sequence of VAV terminal boxes with a reheat coil

3. PROCEDURE FOR THE OPTIMAL MINIMUM AIRFLOW SET POINT

The minimum airflow through terminal boxes is a critical parameter affecting indoor air quality, air circulation and energy consumption. To improve the conventional control sequence, the optimal minimum airflow set point should be determined.

The maximum airflow set point is typically the design cooling airflow, which can be calculated by the room design cooling load.

$$\dot{V}_{max} = \frac{Q_c}{c_p \cdot (T_r - T_{s,clg})} \qquad (1)$$

For VAV reheat terminal boxes that serve exterior zones, the minimum airflow setpoint is typically selected to be the largest of the following:

- The airflow required by the room design heating load, or
- The minimum required for ventilation.

To determine a room design heating load, the building maximum heating load should be calculated. The airflow rate to satisfy the building maximum heating load can be calculated by the following equation:

$$\dot{V}_{\min,h} = \frac{Q_h}{\rho \cdot c_p \cdot (T_{s,htg} - T_r)} \tag{2}$$

The airflow rate that satisfies the outside air ventilation requirements can be calculated by the following equation:

$$\dot{V}_{\min,v} = \frac{\dot{V}_f}{\alpha} \tag{3}$$

where α is the ratio of outside airflow to total airflow. It can be expressed as

$$\alpha = Min\left(\frac{T_R - T_s}{T_R - T_{OA}},\ 1 \right) \tag{4}$$

The optimal minimum airflow set point can be determined by the largest of the previous calculation results between the airflow required by the room design heating load and the minimum required for ventilation.

$$\dot{V}_{\min} = \max(\dot{V}_{\min,h}, \dot{V}_{\min,v}) \tag{5}$$

4. APPLICATION

The applied building is a 5-story office building with a floor area of 91,244 square feet, located in Omaha, Nebraska. There are a total of 106 VAV terminal boxes in this building. Of these boxes, 39 boxes without a reheat coil serve interior zones and 67 reheat boxes serve exterior zones. The boxes are controlled by a pneumatic controller.

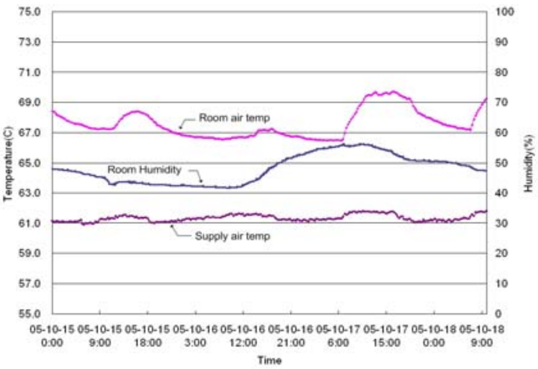
4.1 Conventional terminal box control sequence

Before improving the terminal box control sequence, preliminary investigations were performed by measurement methods intended for all terminal boxes. All thermostat and controller pressure signals were measured with a pressure gauge and regulator. The terminal box airflow was measured using a flow-hood according to the operation of the thermostat. It was found that 79 % of the boxes had inadequate airflow for the conditioned space without considering building operation conditions. The terminal boxes with conventional controls were originally tested and balanced to provide a minimum primary airflow of zero to 100%. These problems may give occupants discomfort and cause excessive energy consumption. Therefore, the terminal box control sequence needed improvement by adjusting airflow to well control in variable air volume (VAV) air-handling unit (AHU) systems. Two typical terminal boxes, located at the interior and exterior zones, were chosen in this study to measure and verify the terminal box control sequence.

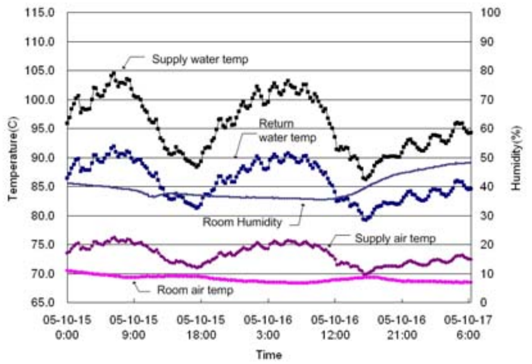
To find the minimum airflow setpoint, the airflow of terminal box in interior and exterior zone was measured using a flow hood. The results are shown in Table 1. The interior and exterior zone terminal boxes were balanced to provide a minimum primary airflow of 25% and 56%, respectively.

Tab.1. Conventional VAV terminal box airflow rate

	CFM_{MIN}	CFM_{MAX}	CFM_{MIN} / CFM_{MAX}
Interior zone box (Without reheat coil)	308 CFM	1199 CFM	25 %
Exterior zone box (With reheat coil)	693 CFM	1234 CFM	56 %



(a)Interior zone



(b) Exterior zone

Fig. 3. Conventional room conditions in interior and exterior zones

Tab.2. Measurement data for conventional room conditions

	Interior zone	Exterior zone
Room air temperature (°F)	66.8 ~ 69.4	67.4 ~ 70.2
Supply air temperature (°F)	61.2 ~ 62.7	70.1 ~ 77.3
Room humidity (%)	42.0 ~ 55.2	37.1 ~ 48.2
Supply water temperature (°F)	-	87.5 ~ 105.2
Return water temperature (°F)	-	79.1 ~ 93.8

To analyze the room air temperature control in an actual system, the room conditions were measured. The range of room air temperature in interior zone was 66.8°F ~ 69.4°F as shown in Figure 3(a) and Table 2. With a conventional minimum primary airflow of 25%, the room air temperature was lower than the set point (72°F). If there was no any cooling load in the conditioned space, the minimum airflow should be zero. Measurement data for the room condition in the exterior zone are shown in Figure 3(b) and Table 2. The conventional minimum primary airflow of 56% gives rise to cold issues. The room air temperature drops lower than that of the set point (72°F). The reason that the room cannot maintain the room air temperature set point is that the supplied primary airflow is higher than the required airflow for the reheat coil. Consequently, the low supply air temperature cannot satisfy the room heating load. Moreover, the high minimum airflow set point often causes significant simultaneous heating and cooling and excessive fan power in mild weather. Therefore, the terminal boxes need to adjust the minimum airflow set point according to building operation conditions.

4.2 Determining minimum airflow set point in an actual system

The minimum airflow set point is determined for exterior zone boxes by applying the previous method to the actual system.

The building load was calculated by the TRACE computer simulation program. The following are input parameters: Double glazing, ¼-inch air space with aluminum frame and thermal break, and then *U* is used as 0.57 Btu/hr·ft^2·°F and the wall *U* is used as 0.05 Btu/hr·ft^2·°F. The floor area is 1650 ft^2, the window area is 287.5ft^2 and the wall area is 402.5 ft^2. The winter outside air design temperature is -8°F, the summer outside air design temperature is 94°F, and the room temperature set point is 72°F. The simulation results show that the cooling load (peak) was 37,560 Btu/hr and heating load (peak) was 16,976 Btu/hr.

The maximum airflow rate in the exterior zone calculated by Equation (1) is 1707 CFM. The minimum airflow for building load calculated by Equation (2) is 377 CFM.

There are three workstations and a research area, so we can assume a maximum occupancy of five working in the area. To meet the ventilation air requirements, we will give 15CFM/person fresh air (ASHRAE Standard 62). Therefore, the fresh air requirement for that area will be 75CFM. During winter time, when outside air temperature is -8°F, the supply air temperature set point is 55°F, and the return air temperature is 75°F, then the AHU outside air intake ratio calculated by Equation (3) is 24%. When the AHU outside air intake ratio is 24%, then the minimum airflow to satisfy the ventilation requirement calculated by Equation (3) is 313 CFM. The minimum airflow selected to be the largest of the previous calculation results by Equation (4) is 377 CFM.

4.3 Improved terminal box control sequence

If there is no cooling load in the conditioned space in the interior zone, the minimum airflow should be zero. The Conventional minimum primary airflow of 25% caused energy waste when there was no cooling load. Therefore, minimum airflow should be reduced from 25% to 0% by adjusting the controller minimum knob to set the minimum airflow.

In the exterior zone, from previous selected minimum airflow, the exterior zone terminal box should be adjusted to 377CFM as the minimum airflow setting, which is about 22% of the maximum airflow set point. Therefore, we can adjust the minimum airflow set point from 56% to 22%. The conventional minimum primary airflow of 56% caused a majority of the rooms to become too cold when the boiler was not operating during the summer months. Where applicable, the minimum primary airflow in the exterior zone was set to 22%. This eliminates the need to operate the boiler during summer months while maintaining the room temperatures at normal levels, which also results in significant fan power and reheat energy savings. By doing this in conjunction with proper static pressure control from the air-handling unit, the rooms receive adequate cooling airflow and do not overwhelm the fan. However, when reduced minimum airflow is less than required, there may be indoor air quality problems and lack of air circulation. Therefore, the room thermal performance should be evaluated after adjusting airflow.

After determining the minimum airflow ratio for each zone, the airflow of the terminal box is measured by a flow hood to verify the minimum airflow ratio. The results are as described in Table 3. The interior zone and exterior zone terminal boxes provide a minimum primary airflow of 0% and 22%, respectively.

Tab 3. Measurement data for iImproved VAV terminal box airflow rate

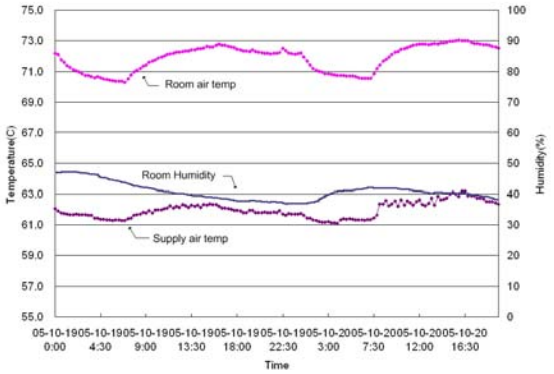
	CFM_{MIN}	CFM_{MAX}	CFM_{MIN} / CFM_{MAX}
Interior Zone (Without Reheat Coil)	0 CFM	1301 CFM	0 %
Exterior Zone (With Reheat Coil)	313 CFM	1213 CFM	22%

4.4 Evaluation of thermal performance

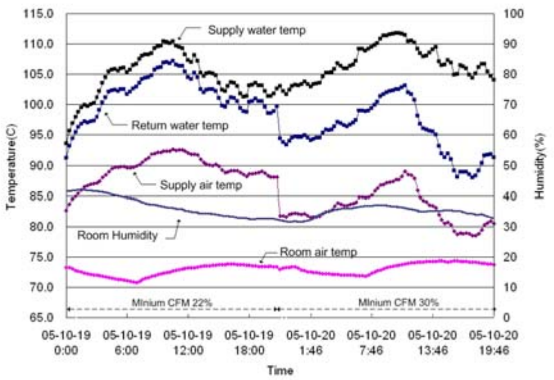
To verify the adjustment of the minimum airflow set point, the room thermal condition was evaluated by room air temperature control, vertical difference of the room air temperature, and indoor air quality.

Stability of room air temperature control

As a result of analyzing the stability of the room air temperature control in an actual system, the range of the room air temperatures in the interior zone is 70.6°F ~ 73.1°F, as shown in Figure 4(a) and Table 4. After adjusting the minimum airflow from 25% to zero, the room air temperature is able to maintain the set point (72°F). Measurement data of the room conditions in the exterior zone are shown in Figure 4(b) and Table 4. The supply air temperature increases and can maintain room air temperature after adjusting the minimum airflow setpoint from 56% to 22%. When adjusting the minimum airflow, the terminal box can control the room conditions. Therefore, it is assumed that there is proper and stable room air temperature control.



(a) Interior zone



(b) Exterior zone

Fig。5. Improved room conditions in interior and exterior zones

Tab. 4. Measurement data of conventional room conditions

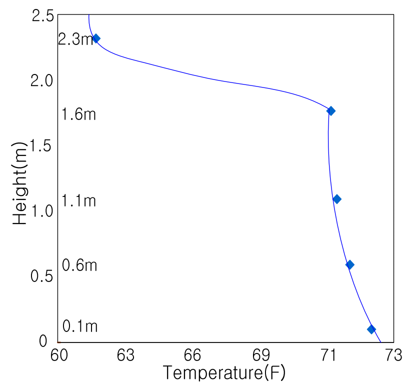
	Interior zone	Exterior zone
Room air temperature (°F)	70.6 ~ 73.1	71.2 ~ 73.5
Supply air temperature (°F)	61.2 ~ 63.3	83.1 ~ 93.3
Room humidity (%)	38.5 ~ 48.1	33.1 ~ 43.2
Supply water temperature (°F)	-	95.7 ~ 113.9
Return water temperature (°F)	-	87.3 ~ 107.8

4.5 Thermal comfort

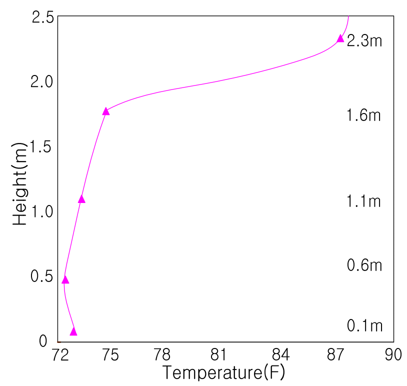
If the minimum airflow is less than required, there may be thermal comfort issues such as indoor air quality problem and lack of air circulation. Therefore, the room comfort issue should be evaluated.

(1) Vertical difference of the room air temperature

To analyze the effects of air circulation in the improved control sequence, measurements were performed by vertical difference of the room air temperature in exterior zone terminal box. In order to measure the vertical distribution of room temperature, five data loggers were installed at intervals of 50cm, at 10cm above the floor. The height where the room air temperature is controlled is 1.1m, the height at which room temperature controllers are generally installed. Figure 5 shows the room air temperature distribution. The cold air and heating air can be seen just under the ceiling. The occupied zone is very nearly uniform temperature with low velocity and should be very comfortable. Therefore, local discomfort due to the vertical difference of room air temperature, when minimum airflow is reduced from 50% to 22%, is not thought to be significant, because the vertical distribution is kept lower than the value proposed by the study on comfort as the vertical temperature difference is below 3ºC (5.4 ºF) between the head and ankles (1.1m and 0.1 m above the floor) (ASHRAE 1992; Olesen, B.W. 2002).



(a) Cooling mode



(b) Heating mode

Fig. 5. Vertical room air temperature in the exteriorzone

(2) Indoor air quality

Measurement of CO_2 in occupied spaces has been widely used to evaluate the amount of outdoor air supplied to indoor spaces. To verify indoor air quality due to reductions in minimum air flow, measurements were performed on the CO_2 levels using indoor air quality meters. To measure CO_2 levels, indoor air quality meters were used in occupied spaces of the interior and exterior zones on each floor. The average CO_2 level on each floor was in the range of 350 ~ 550 ppm. The average outdoor air concentration was 300 ppm. According to ASHRAE Standard 62, comfort criteria are likely to be satisfied if the ventilation results in indoor CO_2 concentrations less than 700 ppm above the outdoor air concentration, which is representative of delivery rates of outside air. Therefore, it is judged that any IAQ problems due to reduction of the minimum airflow set point will not happen.

4.6 Evaluation of energy consumption

Potential energy savings

The reheat energy savings can be considered as the reheat energy consumption required to heat the reduced airflow from the supply air temperature to the room temperature. The reheating coil energy consumption can be estimated based on the following Equation (6).

$$\Delta E_{rh} = \dot{m} \cdot C_p \cdot (\alpha_{\min}{}' - \alpha_{\min}) \cdot (T_r - T_s) \qquad (6)$$

The cooling energy savings equals the cooling energy consumption to cool the reduced airflow from the room conditions to the supply air condition. When the room moisture production is neglected, the potential cooling energy consumption equals the potential reheat energy consumption. The cooling energy consumption can be estimated based on Equation (7) when the outside air temperature is higher than the supply air temperature.

$$\Delta E_c = \dot{m} \cdot C_p \cdot (\alpha_{\min}{}' - \alpha_{\min})(h_r - h_s) \qquad (7)$$

When the outside air temperature is lower than the supply air temperature, mechanical cooling can be eliminated by using an economizer. Therefore, the potential cooling energy savings is zero when an economizer is used. If an economizer is not used, the reduced airflow can decrease the cooling energy consumption even when the outside air temperature is lower than the supply air temperature.

When the impact of the fan efficiency decrease in the base case is neglected, the potential fan power savings can be estimated based on Equation (8):

$$\Delta E_f = \dot{m} \cdot P_{s,d} \cdot (1 - \alpha^2 P_{s,d}) / \eta_{f,d} \qquad (8)$$

When the supply fan airflow is reduced, the supply fan speed should be reduced and the total fan power will be dropped from Equation (8). Therefore, the significant fan power should be saved, and with proper static pressure control from the air handling unit, the rooms receive adequate cooling airflow and do not overwhelm the fan. In addition, if the water flow rate in the reheat coil is reduced from adjusting the airflow, the potential pump power should be saved. Consequently, the energy consumption should be optimized between the airflow and reheat energy.

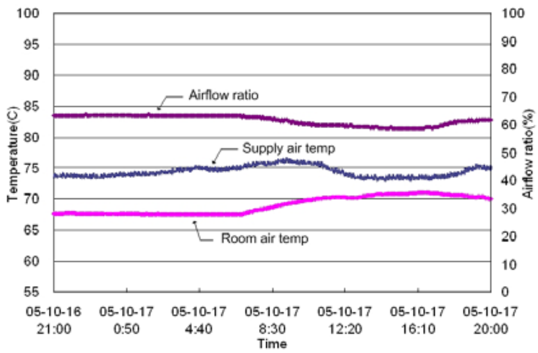
Energy consumption in the actual system

In order to evaluate the thermal energy consumption when optimizing the minimum airflow to the exterior zone terminal box, thermal energy is compared between conventional minimum airflow and improved minimum airflow. To measure the airflow of the terminal box in the exterior zone, an air velocity meter was installed in the duct. As shown in Figure 6, the conventional minimum airflow ratio is 56% of the primary airflow. The room air temperature cannot maintain the room set point (72F) because too low supply air temperature cannot eliminate the heating load. The improved minimum airflow indicates that 22% of primary airflow and a suitable supply air temperature can maintain the room air temperature.

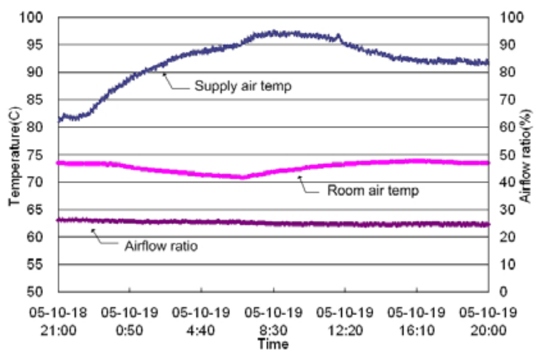
The thermal energy consumption can be calculated with Equation (9):

$$Q_e = \dot{m} \cdot c_p \cdot (T_{S,aB} - T_{S,bB}) \quad (9)$$

When the outside air temperature is 55°F, the energy consumption is 8177 Btu/hr when there is conventional minimum airflow (56%), as shown in Figure 7. On the other hand, the energy consumption is 4623 Btu/hr when there is improved minimum airflow (22%). The thermal energy consumption of improved minimum airflow is less than that of the conventional minimum airflow by 43 %. Therefore, according to adjustment of the minimum airflow, the terminal box in exterior zone can reduce thermal energy.



(a) Conventional condition



(b) Improved condition

Fig. 6. Airflow ratio in conventional and improved conditions

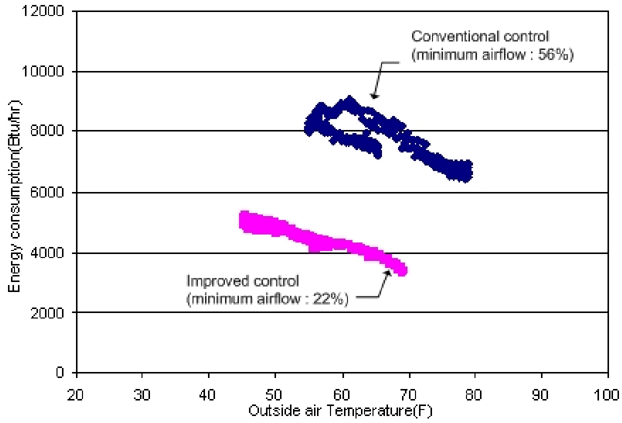


Fig.7. Comparison of the thermal energy consumption in exterior zone

5. CONCLUSIONS

To improve the conventional terminal box control sequence, control factors and methods were analyzed in this study. A suitable airflow was selected, and the applicability of improved sequence was estimated through the measurement and verification method. Energy performance was evaluated through simulation. The results are as follows.

(1) In conventional control, the terminal boxes were originally tested and balanced to provide a minimum primary airflow of zero to 100%. The room temperature could not maintain the set point because the minimum airflow supplies an inadequate airflow for a conditioned space without considering building operation conditions. The minimum airflow is higher than required, which often leads to significant simultaneous heating and cooling, in addition to excessive fan power. Therefore, the terminal boxes need to adjust their minimum airflow setpoint according to building operation condition.

(2) In improved control, adjustments to the minimum and maximum primary airflows were made. Where applicable, the minimum primary airflow in interior zone was set to 0% and exterior zone was set to 22%. This gives thermal environmental comfort in the conditioned area, and also results in significant thermal energy savings.

(3) Through the measurement and verification method, improved control can stably maintain the set room air temperature and reduce energy consumption, compared to the conventional control. Moreover, the vertical difference in room air temperature satisfies within comfort range. Measurements of CO_2 levels show there is no indoor air quality problem when the minimum airflow set point is reduced.

(4) In energy measured results, the thermal energy consumption with improved minimum airflow is 43% less than that with the conventional minimum airflow. Therefore, if adjustments are made in minimum airflow, a terminal box in the exterior zone can reduce thermal energy.

NOMENCLATURE

c_p - specific heat capacity, Btu/lbm °F

ΔE_f − fan power, kw

h_r - room enthalpy, Btu/lbm

h_s - supply enthalpy, Btu/lbm

$\dot{m}$ - mass flow rate, lbm/hr

$P_{s,d}$ − the static pressure set point, in.wg

T_m - mixed air temperature, °F

T_{OA} - outside air temperature, °F

T_R - return air temperature, °F

T_r - room dry bulb temperature, °F

T_s - supply dry bulb temperature, °F

$T_{s,aB}$ - supply dry bulb temperature for after terminal box, °F

$T_{s,bB}$ - supply dry bulb temperature for before terminal box, °F

$T_{s,\mathrm{clg}}$ - supply dry bulb temperature for cooling, °F

$T_{s,htg}$ - high limit of the supply dry bulb temperature for heating, °F

Q_c - cooling design load of the room, Btu/hr

Q_h - heating design load of the room, Btu/hr

Q_e − thermal energy consumption of the room, Btu/hr

$\dot{V}_{\max}$ - maximum air volumetric flow rate supplied to the room, ft^3/min

$\dot{V}_{\min,h}$ - minimum air volumetric flow rate supplied to the room, ft^3/min

$\dot{V}_{\min,v}$ - minimum air volumetric flow rate supplied to the room for ventilation, ft^3/min

$\dot{V}_f$ - air volumetric flow rate for fresh air requirement, ft^3/min

α − AHU outside air intake ratio, %

$\alpha_{\min}'$ − minimum airflow ratio in conventional conditions

$\alpha_{\min}$ − minimum airflow ratio in improved conditions

ρ - standard air density, lbm/ft^3

$\eta_{f,d}$ − fan efficiency

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